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Simulation of Crankshaft Torsional Vibration by Flexible-body Dynamics

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Abstract

The crankshaft accidents due to torsional vibration become severe with increasing number of rows of reciprocating compressors. In order to calculate more accurately of the crankshaft torisional vibration, a new calculation method based on flexible multi-body dynamics theory was presented in this paper. The model establishment process was introduced and the torsional vibration of a 6 rows crankshaft before and after transformation was simulated by ADAMS software. The results showed that before transformation the natural frequency of the crankshaft was closed to the excitation frequency, thus caused the vibration amplitude of crankshaft too large. Actual stress of the piston pin of the first and the second row increased rapidly. After transformation, the frequency of the crankshaft became away from the excitation frequency, and the amplitude of the crankshaft torsional vibration reduced substantially, then the crankshaft accident disappeared.

Key words: crankshaft, torsional vibration, ADAMS, reciprocating compressor

1.Introduction

The reciprocating compressor plays an increasingly important role in the modern petrochemical industry. With increasing requirements of gas with large mass flow and high pressure, the demand of large-scale multi-row reciprocating compressor is rapidly expanding. It also brings many problems, such as the torsional stiffness of the crankshaft becoming lower while the inertia of the compressor becoming larger. Thus the crankshaft natural frequency is closed to the excitation frequency, and the crankshaft face to great risk of torsional resonance. In practical engineering, the compressor fault and the crankshaft fracture frequently occur due to crankshaft torsional vibration. Therefore, the crankshaft torsional vibration has become a key point of the development of the large-scale reciprocating compressors.

Currently, domestic and foreign scholars on the crankshaft torsional vibration did a lot of research and made a great progress. In domestic traditional calculation method of the crankshaft torsional vibration, the result is not accurate enough, because the inertia of the reciprocating mass is thought as a constant, while it changes with the rotational angle. In order to improve the accuracy of the traditional calculation method, a new simulation method for the crankshaft torsional vibration was proposed in this paper.

In this paper, we choose the 6M50 reciprocating compressor with 6 rows and the maximum piston force of 50 tons as an example to study. The crankshaft material is 35CrMo, and it has 6 cranks. Each cylinder was arranged in one row, as shown in Fig. 1.Table 1 shows the main technical parameters of the compressor.

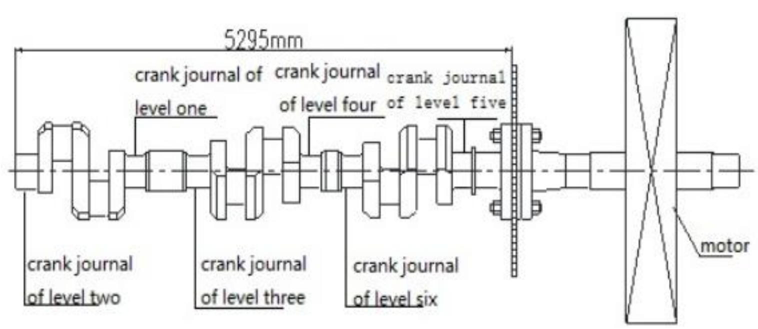


Fig. 1 The structure of compressor crankshaft

Table. 1 The main technical parameters of 6M50 compressor

List	Value
The number of rows	6
Suction pressure(*MPa*)	0.1
discharge pressure(*MPa*)	31.4
The stroke of pistons(*m*)	0.45
Rotational speed($r \cdot min^{-1}$)	300
Crank throw(*m*)	0.175
The length of the connecting rod(*m*)	0.95
The maximum gas force(*kN*)	380
The maximum piston force(*kN*)	363
The diameter of the main shaft(*mm*)	300
The cylinder diameter of level one(*mm*)	1390

According to the accident statistics of the compressor company, the average operating rate is only 48.8% during the four years' operation. The major accidents happened on the crankshaft are the crank pins of the first row and the second column scraping frequently, and the average life of them is only 202h and 304h respectively. During 7,028h's running time, there are three crankshafts cracked. Two of the them cracked up to 32mm in depth of the second crank pin. In order to eliminate the accidents, the company made a new crankshaft from the two waste crankshafts, in which the first two rows were connected with the other four row-by a rigid coupling, as shown in Fig. 2.The accidents disappeared since the new crankshaft was used. The compressor operates normally by far.

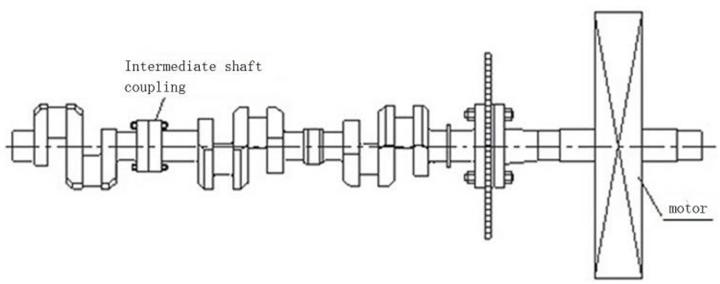


Fig .2 the crankshaft structure after transformation

To analyze the crankshaft accidents, this paper proposed a new calculation method of the crankshaft torsional vibration based on flexible multi-body dynamics theory. The new crankshaft and the old one were both simulated by ADAMS respectively. then the reasons of the crank pin failure were also analyzed.

2 Finite element model of the crankshaft on the foundation of Flexible-body Dynamics

Firstly, we built a 3-D solid model in 3-D software, as shown in Fig. 3. Because fillets and oil holes on the crankshaft do not affect the crankshaft torsional vibration characteristics, so they can be neglected.

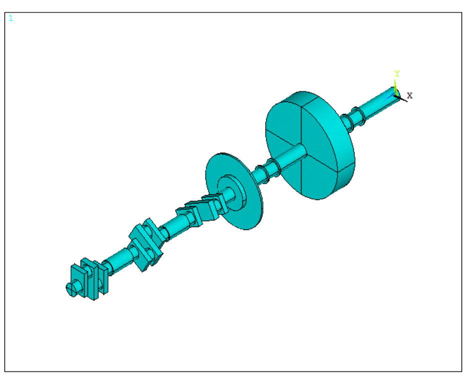


Fig. 3 3-D model of the crankshaft

And then the 3-D solid model was imported into ANSYS software, and physical parameters of the crankshaft were input. The crankshaft model is divided into 99519 elements of SOLID73. At last a MNF file was output to ADAMS for torsional vibration model establishment, as shown in Fig. 4.

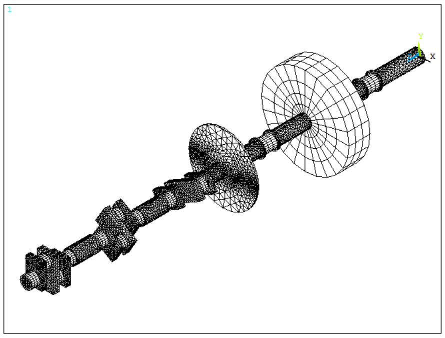


Fig .4 3-D finite element model of the crankshaft

This MNF file contains information of the mass of flexible body, center of mass, moment of inertia, frequency, and vibration modes etc. In ADAMS the crankshaft is connected with the connecting rod, cross head, piston rod and the piston of the compressor, and then the load movement and boundary conditions to the model were applied, as shown in Fig. 5.

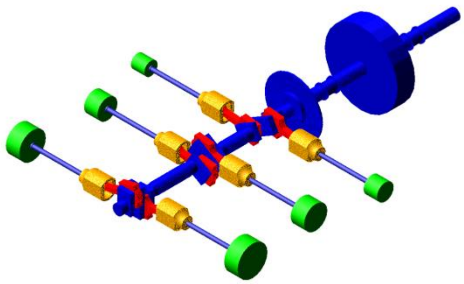


Fig. 5 crankshaft torsional vibration model

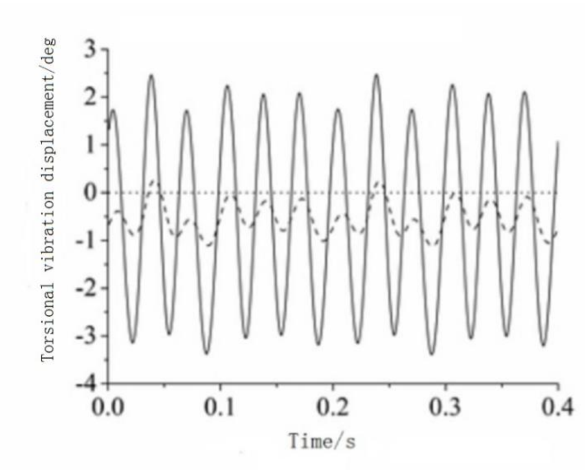
The moving boundary condition is the motor speed. The gas force is loaded by STEP function. The STEP function as follows

$$STEP(VX_i, 0, STEP(P_{out} - P_{in} \cdot ((S+S_0)/(X_i+S_0))^{\kappa_2}, 0, P_{out}, 1e-4,$$
$$P_{in} \cdot ((S+S_0)/(X_i+S_0))^{\kappa_2}), 1e-4, STEP(P_{out} \cdot (S_0/(X_i+S_0)^{\kappa_1} - P_{in}), 0, P_{in}, 1e-4,$$
$$P_{out} \cdot (S_0/(X_i+S_0)^{\kappa_1}))$$

In this paper, the GSTIFF integrator is chose, the time step is 0.001*s*, and the accuracy requests is 0.01.

3 Analysis of Torsional Vibration Characteristic

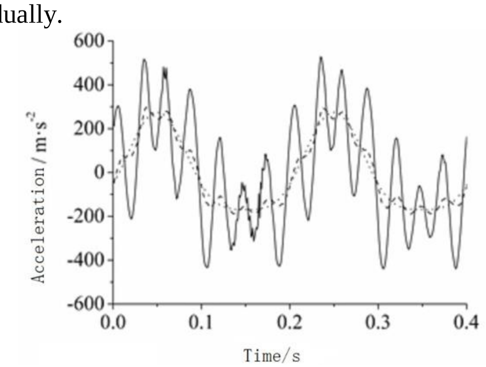
As mentioned above, the crankshaft transformation actually is adding a flywheel between 2# and 3# main bearing. In this paper, simulations are done on practical running process of the new crankshaft and the old one respectively. The crank pin of the first row is chosen as an example. Fig. 7 shows the torsional vibration displacement of the new crankshaft and that of the old one. It can be seen that the crank pin torsional vibration of the new crankshaft decreased greatly. The maximum torsional displacement of the first row crankpin of the old crankshaft is about -3.30deg, while that of the new crankshaft decreases to -1.04deg, reduced by 68%.



—— old crankshaft - - - - new crankshaft

Fig. 7 torsional vibration displacement of the first row crankpin

Fig. 8 shows the first row piston acceleration curve. It can be seen that the new crankshaft fluctuation value has greatly reduced, which is almost similar to the rigid body dynamics. The maximum fluctuation value of old crankshaft is 352$m \cdot s^{-2}$. However, the maximum fluctuation of the new crankshaft value reduces to 54.6$m \cdot s^{-2}$,reduced by 84.8%.Variation trends of the other five rows are similar to the first row, and the fluctuation value from the first to the sixth row decreases gradually.



——old crankshaft ------- new crankshaft

Fig. 8 acceleration of the first row piston

4 Comparison of the simulation results

To analyze the simulation results, Fourier decomposition is used to the crankshaft torsional vibration displacement .Because the variation trend of each crank pins torsional vibration displacement is consistent, so only the first row crank pin is decomposed as an example, as shown in Fig. 9.

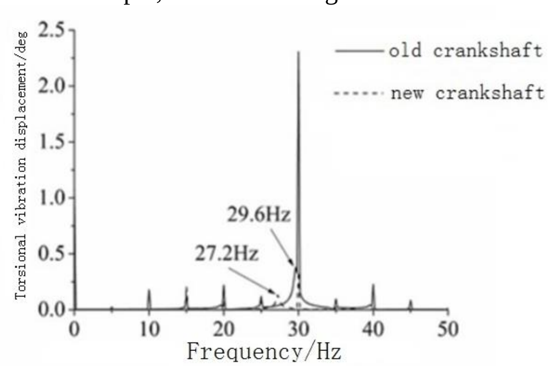
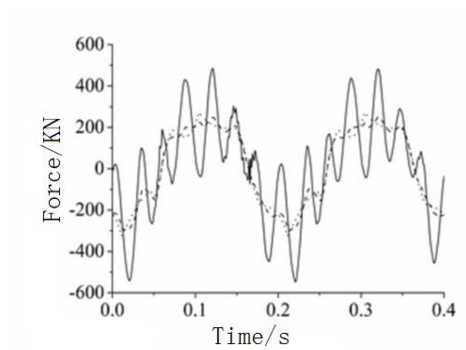


Fig. 9 Fourier decomposition of the first row crank torsional vibration displacement

From Fig. 9 it can be seen that the crankshaft torsional vibration has a peak in 5 Hz and its integral multiples. These peaks are the comprehensive effect of the each force results. The peak of the new crankshaft decreases from 29.6 *Hz* to 27.2 *Hz*. Because the 29.6 *Hz* is close to 30 *Hz*, the old crankshaft has a higher peak at 30Hz, the torsional vibration displacement is up to2.3deg, which causes the crankshaft's large amplitude and running erratic. However, the new crankshaft natural frequency is far from 25 *Hz* and 30 *Hz*, so the torsional vibration amplitude of crankshaft is slower than that before, and the compressor is running more stably.

5 Failure Analysis of Crankpin Bearing

By simulation, the piston pin force along the cylinder center line is obtained, as shown in Fig. 10. Because the crankshaft torsional vibration deformation is ignored, the calculation result of the rigid body dynamics is the design value.



—— old crankshaft ---- new crankshaft

Fig. 10 The piston pin force along the cylinder center line

It can be seen from Fig. 10 that the piston force of the old crankshaft has a large torsional vibration amplitude. In the compressor, reciprocating mass of the low stage at the first and the second rows are very large, and the piston pin force of these stage could exceed the design value. The maximum value of the piston pin force occurred in the second row is 483.4*kN*. However, its design value is only313*kN*. These data indicate that the large amplitudes of the crankshaft lead to the piston pin force's increasing, while the torsional vibration amplitudes of the new crankshaft decreases, and the piston force of the each row is agreed with the design value.
Table2 and table 3 respectively show the maximum pressure ratio k_{max} and the impact coefficient *S* of the connecting rod bearing.

Table 2 The maximum crank pin bearing pressure ratio of each row of 6M50 compressors

	First column	Second column	Third column	Fourth column	Fifth column	Sixth column
Design value(MPa)	8.3	7.8	6.6	7.9	9.2	10.6
Old crankshaft(MPa)	15.2	14.5	7.9	8.4	10.4	11.1
New crankshaft (MPa)	9.0	8.7	6.8	8.0	9.3	10.7

Table 3 The crank pin bearing impact coefficient of each row of 6M50 compressor

	First column	Second column	Third column	Fourth column	Fifth column	Sixth column
Design value	1.7	1.6	1.6	1.6	1.7	1.9
Old crankshaft	2.9	2.8	2.0	1.8	1.9	2.0
New crankshaft	1.8	1.8	1.7	1.7	1.7	1.9

According to the technical manual of the displacement compressors, the maximum value of k_{max} and *S* are 9 MPa and 2.5 MPa respectively. These data indicate that the old crankshaft piston pin stress in the first and second row increase rapidly, and the impact factor of the crank pin bearing has reached to the upper limit. So the oil film rupture easily happens, and causes crank pin bearing abrasion and failure. While the new crankshaft natural frequency is far away from the excitation frequency. So the vibration amplitude of the crankshaft decreased substantially, and the maximum pressure ratio k_{max} and impact coefficient *S* are all in the scope of design.

6.Conclusion

This paper proposed a new calculation method based on the flexible multi-body dynamics theory. According to this method, the torsional vibration characteristic of a 6 row crankshaft were calculated. The results show that torsional natural frequency of the crankshaft before transformation is close to the excitation frequency, so the crankpins of the first and the second row often cracks because of the torsional vibration. After transformation, the new crankshaft natural frequency is away from the excitation frequency, and the vibration amplitude of the crankshaft decreases substantially. The maximum pressure ratio k_{max} and impact coefficient *S* are all in the scope of the safe design, so the crankshaft accident disappears.

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